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EFFECT OF LOAD DIRECTION ON THE OIL FILM TEMPERATURE DISTRIBUTION OF 3-LOBE PERICYCLOID JOURNAL BEARING

VLIV SMĚRU ZATĚŽOVÁNÍ NA DISTRIBUCI TEPLOTY V OLEJOVÉM FILMU TROJLALOKOVÉHO PERICYKLOIDÁLNÍHO RADIÁLNÍHO LOŽISKA

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Abstract

The 3-lobe journal bearings are applied in the bearing systems of rotating machinery. The design of bearing, the number of lobes and oil grooves allow for good cooling and very good stability of bearing. The 3-lobe journal bearings can be manufactured as the bearings with cylindrical operating surfaces as well as the bearings with the pericycloid profile of bearing bore. The classic 3-lobe journal bearing is designed as the bearing of cylindrical, non-continuous profile but the 3-lobe pericycloid bearing has the continuous profile of bore.

The paper presents an effect of load direction on the oil film temperature distribution of 3-lobe pericycloid journal bearing. The oil film pressure, temperature and viscosity fields were obtained by iterative solution of the Reynolds', energy and viscosity equations. Laminar adiabatic oil film, static equilibrium position of journal and parallel axis of journal and bearing were assumed. The temperatures of oil film were verified experimentally.

Key words: load direction, oil film, 3-lobe journal bearings

1. Introduction

The multilobe bearings, mostly used in the slightly loaded, high-speed machines are characterised by good cooling conditions, good damping of vibrations and the stable operation [1-3]. Their geometry is introduced in the literature but there is a lack of information on the static and dynamic characteristics of three-lobe bearings with the pericycloid profile [1-3]. Such bearings are used, e.g. in the bearing system of the grinders spindles or in the high-speed compressors. Typical three-lobe journal bearing is composed of single circular sections whose centres of curvature are not in the geometric centre of the bearing. The geometric configuration of the bearing as a whole is discontinuous and not circular. The three-lobe pericycloid journal bearing (wave bearing) [2, 4] is characterised by continuous profile and three hydrodynamic oil films on the journal perimeter. Pericycloid is a continuous curve that is a trajectory of plane point of circle undergoing pure rolling with internal curvature on a fixed circle. Continuous curvature of the operating surface is an important feature of the pericycloid bearing. Such a configuration allows simultaneous machining of the whole surface by simple workshop techniques and hence precise shape as well as the dimensional accuracy.

The operating characteristics of the bearing consist among others the pressure, temperature distribution in oil film.

The properties of the finite length pericycloid bearing have been partly investigated for isothermal and adiabatic oil film and the numerical method employed has allowed a comprehensive calculation of the static and dynamic characteristic. Experimental investigations of finite pericycloid bearings have confirmed their advantages in both characteristics and in the practical application in high speed bearing in the range of laminar oil film.

The paper presents the results of the calculations of oil film temperature distribution of 3-lobe journal bearing with the pericycloid profile on the assumption of different load direction. The Reynolds, energy and viscosity equations were solved numerically on the conditions of static equilibrium position of the journal. The temperatures of oil film were measured on the operating surfaces of pericycloid journal bearings applied in the grinder spindle. Such investigation are the first that were carried out on the real bearing system with pericycloid profile bearings.

2. Oil film geometry

The geometry of the oil film gap of multilobe journal bearing [2, 4] is generally described by Eqn (1) at the assumption of parallel axis of journal and sleeve.

$$\overline{H}(\varphi,z)=H_c+\overline{H}_{P,L}(\varphi) \tag{1}$$

The first member of right side of Eqn. (1) that gives the oil gap thickness for eccentric orientation of journal in the bearing bush has the following form:

$$H_c=1-\varepsilon\cdot\cos(\varphi-\alpha) \tag{2}$$

where: ε- relative eccentricity, φ - peripheral co-ordinate, α - attitude angle of centres line.

The oil film thickness of the pericycloid bearing [2, 4] at concentric position of journal and bush can be expressed in non-dimensional terms as:

$$\overline{H}_P=\lambda^*\,(1+\cos n\,\varphi) \tag{3}$$

where: λ^* - relative eccentricity of pericycloid, n – multiply of pericycloid.

3. Oil film temperature distribution

The oil film pressure, temperature and viscosity distributions have been determined by means of Reynolds, energy and viscosity equations [1-4]. These equations were derived on the assumption that in the bearing gap exists the adiabatic laminar flow of non-compressible Newtonian fluid. The oil film pressure distribution was calculated from the following Reynolds equation:

$$\frac{\partial}{\partial\varphi}\left(\frac{\overline{H}^3}{\overline{\eta}}\frac{\partial\overline{p}}{\partial\varphi}\right)+\frac{\partial}{\partial\overline{z}}\left(\frac{\overline{H}^3}{\overline{\eta}}\frac{\partial\overline{p}}{\partial\overline{z}}\right)=6\frac{\partial\overline{H}}{\partial\varphi}+12\frac{\partial\overline{H}}{\partial\phi} \tag{4}$$

where: $\overline{p}$ -dimensionless oil film pressure, $\overline{\eta}$ - dimensionless dynamic oil viscosity, ϕ - dimensionless time, $\overline{z}$ -dimensionless axial co-ordinate

The oil film temperature distribution equation has the following form

$$\begin{aligned}&\frac{12\cdot\overline{H}}{Pe}\left(\frac{\partial^2\overline{T}}{\partial\varphi^2}+\left(\frac{D}{L}\right)^2\frac{\partial^2\overline{T}}{\partial\overline{z}^2}\right)+\left(\frac{\overline{H}^3}{\overline{\eta}}\frac{\partial\overline{p}}{\partial\varphi}-6\cdot\overline{H}\right)\cdot\frac{\partial\overline{T}}{\partial\varphi}+\left(\frac{D}{L}\right)^2\frac{\overline{H}^3}{\overline{\eta}}\frac{\partial\overline{p}}{\partial\overline{z}}\cdot\frac{\partial\overline{T}}{\partial\overline{z}}=\\&-\frac{\overline{H}^3}{\overline{\eta}}\left(\left(\frac{\partial\overline{p}}{\partial\varphi}\right)^2+\left(\frac{D}{L}\right)^2\left(\frac{\partial\overline{p}}{\partial\overline{z}}\right)^2\right)-\frac{12\cdot\overline{\eta}}{\overline{H}}\end{aligned} \tag{5}$$

where: $\overline{T}$ - dimensionless oil film temperature, Pe -Peclet number,

Oil film pressure, temperature and viscosity fields can be obtained by numerical solving of Eqn. (1) through Eqn. (5). The pressure boundary condition assumes the positive values of oil pressure only and the ambient pressure on the sides of bearing. It was assumed that in the regions of negative pressures $p(\varphi,\overline{z})=0$. The temperature $T(\varphi,\overline{z})$ on the bearing edges ($\overline{z}=\pm 1$) was computed by the method of parabolic approximation [2-4].

4. Results of calculations

The program of numerical calculations [4] of multilobe bearings with laminar or turbulent oil film was applied. Bearing length to diameter ratio L/D=0.8, clearance ratio ψ=2,8‰ , ψ=0,4‰ , the pericycloid eccentricity $\lambda^* = 0.2$ were considered. The values of heat number K_T was calculated on the assumption of journal diameter d= 80 mm, bearing minimum clearance ΔR_{min} = 0,05 mm and relative clearance ψ=0.4 ‰ .

For assumed values of relative eccentricities ε, the oil film temperature distributions, the resultant oil film forces So, static equilibrium position angles α_{eq}, oil film maximum temperature T_{max} were calculated. Exemplary results of calculations are shown in Fig. 1 through Fig. 3. An effect of load direction on the oil film temperature is presented in Fig. 1. The load direction was modelled by the arrangement of bearing operating surfaces with regard to the vertical load that was directed down. There is the difference in the oil film temperature distributions for the considered cases of load direction (Fig. 1).

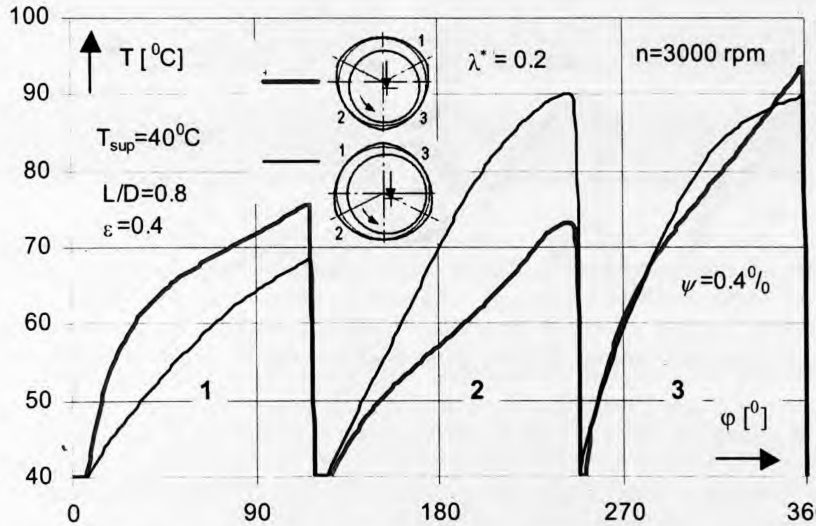
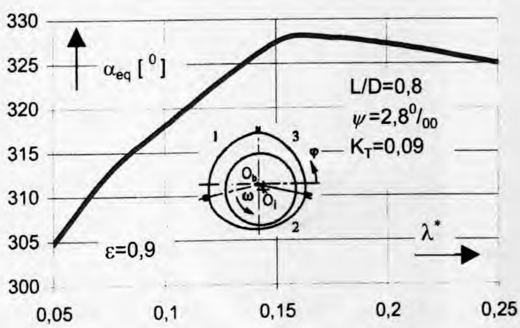


Fig. 1 Oil film temperature distribution in pericycloid 3-lobe journal bearing at different load angles



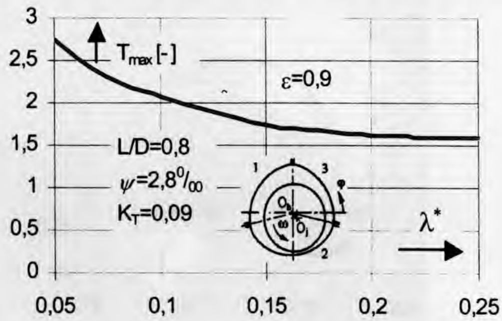


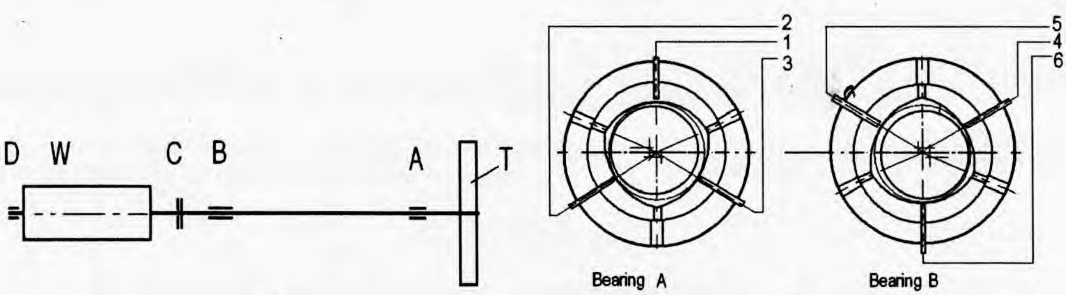
Fig. 2 Effect of pericycloid eccentricity λ^* the static equilibrium position angle

Fig. 3 Effect of pericycloid eccentricity on on the oil film maximum temperature

An effect of pericycloid relative eccentricity λ^* on the static equilibrium position angle α_{eq} is showed in Fig. 2; an increase in the pericycloid relative eccentricity causes an increase in the static equilibrium position angles α_{eq} to about relative eccentricity $\lambda^* = 0.16$ and the decrease at higher values of pericycloid eccentricity. The maximum temperature of oil film shows the decrease at the increase in the values of pericycloid relative eccentricity (Fig. 3).

5. Experimental investigation of the oil film temperatures of bearings

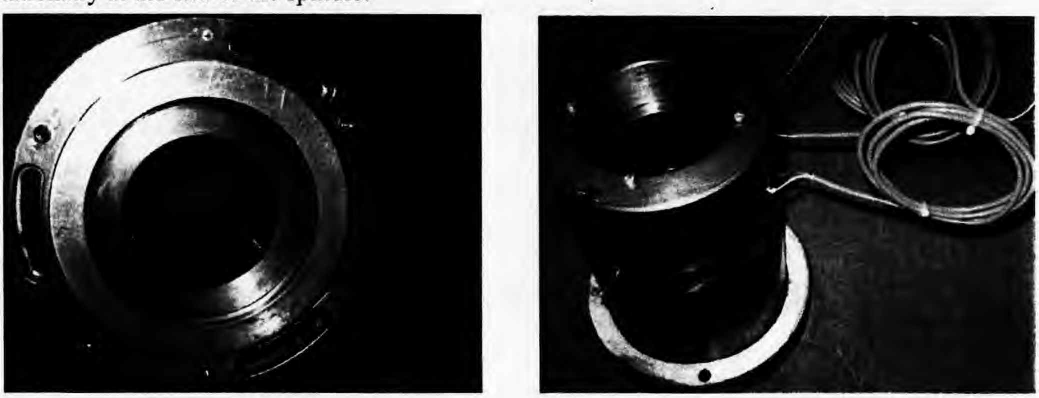
The test rig for the investigation of oil film temperatures is based on the grinder spindle assembly. The system of the journal bearings of grinding spindle is showed in Fig. 4a. Each bearing was equipped in three doubled thermocouples NiCr-NiAl placed in the center of bearing lobe about 0.2 mm below its surface (Fig. 4b). Two additional thermocouples were applied for the measurements of temperatures of bearing casings. The bearings system was lubricated with the oil Velol 9 of the kinematic viscosity 9-11 mm^2/sec at 40^0C supplied from the central lubricated system. The acquisition system that was coupled with the computer has allowed for the continuous measurements of temperatures during assumed period of time. The rotational speed of spindle was 1500 rpm at the rated power of electric motor 7.5 kW.



a) b)

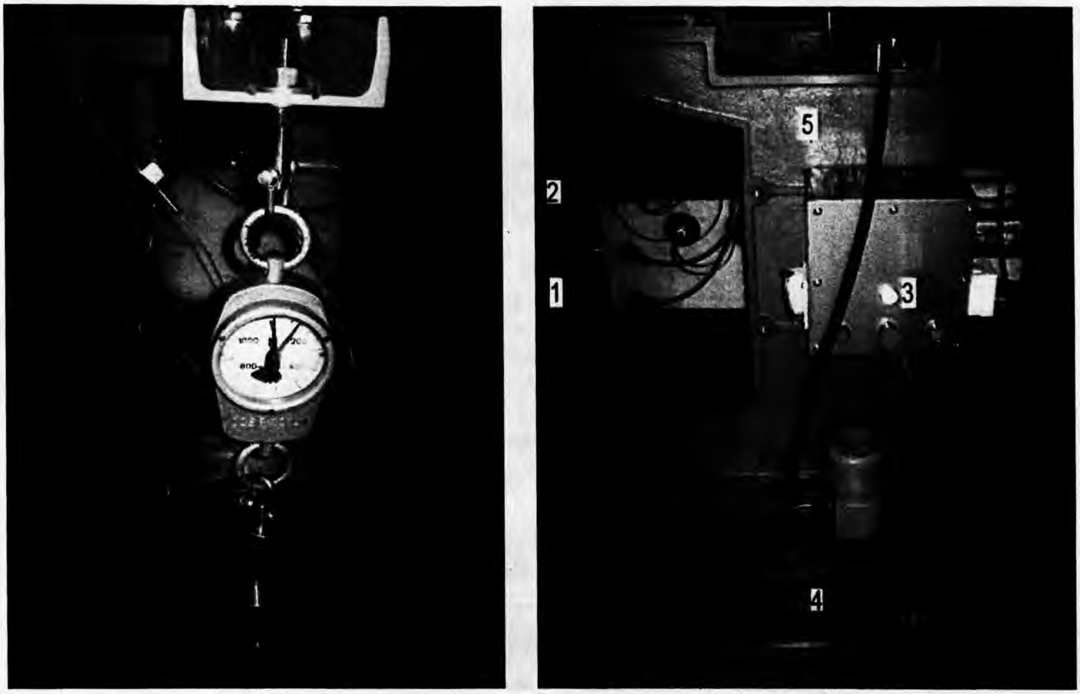
Fig. 4 Journal bearings system of the spindle of grinding machine; A, B – pericycloid 3-lobe journal bearings, C- axial ball bearing, D- deep groove ball bearing, W – rotor of electric motor, T –grinding wheel

The casings of bearings with the sleeves made of bronze are showed in Fig. 5. General view of test rig with the loading device is presented in Fig. 6; the vertical load in the upper direction was obtained by means of tightening the bolts fixed to the dynamometer; the vertical load in the opposite (down) direction was obtained by the weight fixed directly to the casing which was mountad rotationally at the end of the spindle.



a) b)

Fig. 5 Bearing casings and sleeves with the thermocouples cables; a- bearing A, b- bearing B



a) b)

Fig. 6 Test rig with the grinder spindle operating in pericycloid 3-lobe journal bearings and the loading system; a - side view; 1 - spindle casing, 2 - spindle, 3 - control system, 4 - pressurized oil supply system, 5 - frame of test rig, b - face view (loading system)

The measured values of temperatures were compared to the calculated ones at the assumed conditions of oil supply. Good agreement of theoretical and experimental results was obtained.

Exemplary values of measured temperatures on the operating surfaces of bearings are showed in Fig. 7. Total time of the measurements was 8800 sec. The temperatures were registered each 5 sec but the values that are given in Fig. 7 are the mean values from 8 measurements in each series. The highest temperatures were obtained on the lobes No. 4 and No. 5 of bearing B (Fig. 4), the lowest ones on the lobes No. 1 and No. 3 of bearing A (Fig. 4) and the middle temperatures were measured on the lobes No. 2 and No. 6 of respective bearings (the temperatures on the lobe No. 6, bearing B are slightly higher than on the lobe No. 2 of bearing A).

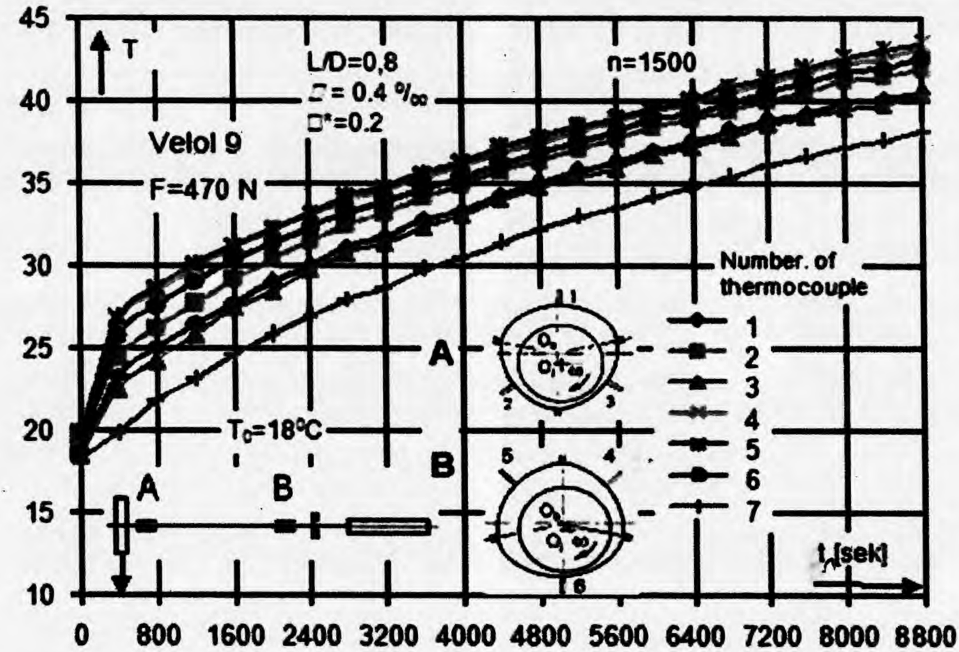


Fig. 7 Measurements of the oil film temperatures at the vertical load 470 N; bottom curve shows the temperature of the casing of bearing A (thermocouple No. 7)

6. Final remarks

The paper summarizes the results of the calculations and experimental investigation of oil film temperatures of pericycloid journal bearings that were applied in the real mechanical system of grinding spindle. The calculated temperatures were obtained for single bearing. In real bearing system, the temperatures of oil film are affected by many factors, e.g. casing design, temperature of supplied oil and its pressure. However, the results obtained from the theory and experiment show the correct values on the bearing lobes. In calculations and experiment different directions of journal rotation were considered but there is good agreement in, e.g. maximum oil film temperatures on the operating lobes of bearings. At anticlockwise direction (theory) of journal rotation and for bearing A and B, the maximum temperature was observed on the lobes No. 3 and No. 2 in the bearings A and B, respectively (Fig. 1). These temperatures correspond to the measured temperatures (at clockwise direction of journal rotation) that occur on the lobe No. 2 (bearing A) as well as No. 4 and No. 5 (bearing B) at clockwise direction of journal rotation (Fig. 7). As the final remark it can be concluded that the numerical calculations and experiment have confirmed the possibility of obtaining the real values of oil film temperatures.

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